CHAPTER 13

Fatigue and Fracture Mechanics

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CHAPTER 13

Fatigue and Fracture Mechanics

Introduction

1 The term 'fatigue' is used to describe the failure of a material under a repeatedly applied stress less than the ultimate strength of the material. 'Fracture mechanics' is a generic term covering the application of analytical and experimental methods to predict the behaviour of cracks in specimens and structures. Phenomena which have been modelled by fracture mechanics methods include brittle fracture, tearing, fatigue crack growth, stress corrosion cracking and creep. Unlike other chapters in this publication, the theory of fracture mechanics will not be summarised here as it is almost impossible to present it in a shortened form while at the same time keeping it useful and understandable. It is not particularly complicated, but it does require some dedicated study, and one of the best introductory texts is that of Rolfe and Barsom (1977), whilst for fatigue Rollason (1977) and Gurney (1979) are recommended. This chapter, however, aims to provide the structural designer with simple and quick guidance on how to assess the susceptibility of a ship's structure to fatigue failure and the risk of fast fracture both during design and in service.

2 For ship structures the most likely damage mechanism is fatigue. In mild steel to BS4360 grade 43A brittle fracture could occur as a result of collision, severe slamming or weapon attack as discussed by Sumpter et al (1989). In tougher grades of steel, damage from these events is likely to be confined to local tearing. Experience shows that, even in grade 43A steel, fracture under normal wave loading is unlikely, except at extreme combinations of low temperature, large crack size and high residual stresses adjacent to a weld. Fatigue on the other hand has been a frequently encountered problem. Fortunately, fatigue cracks advance in a relatively slow manner and are generally a nuisance rather than a serious risk to structural integrity. Ultimately, however, a long fatigue crack could cause failure of a ship's hull, and the best practice, as emphasised in this chapter, is to set stress levels so as to avoid the onset of fatigue cracking in all primary or critical structure.

Use of S-N Curves

3 The basic principle in design against fatigue is the use of S-N (or σ-N) curves which relate the tensile stress *range* (S or σ) to the number of cycles (N) of that stress to initiate a fatigue crack; the stress is known as the *endurance limit* at N cycles; a typical set of fatigue curves is shown in *Figure 13.1*. Many tests in a large number of engineering materials have been carried out to establish fatigue curves and are readily available, for example Boyer (1986) as well as BS5400

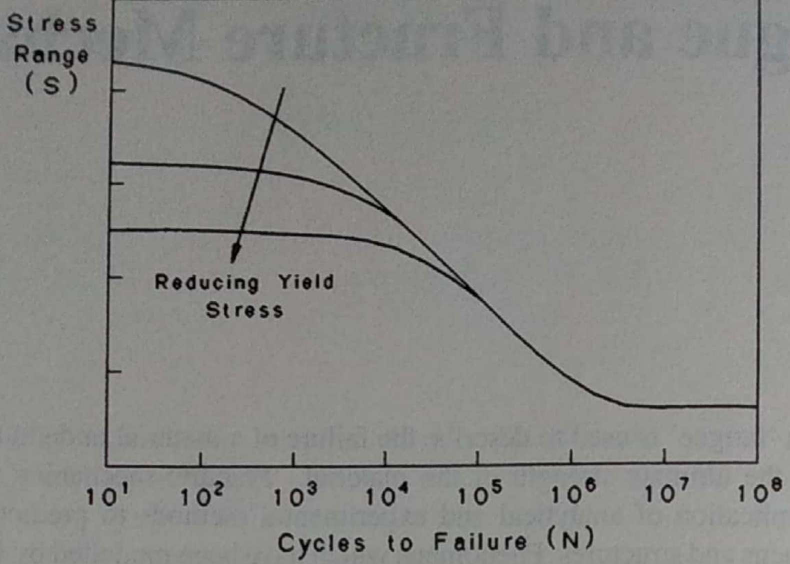

Figure 13.1 Typical Fatigue S-N Curve

Part 10 for structural steels and BS8118 for aluminium. For ferritic steels below 200°C, titanium and its alloys, magnesium and some aluminium-magnesium alloys at room temperature the S-N curves tend to become horizontal after about 10^7 cycles at a stress range which is called the *fatigue limit*. With most non-ferrous metals and austenitic steels no definite fatigue limit is found, neither does there appear to be a fatigue limit in complex welded steel structures or for steel in the presence of seawater, even if the parent material exhibits such a limit. This is illustrated in the design curves given in BS5400 Part 10. It should be noted also that there is always considerable scatter in the endurance limit values obtained by experiment and so the use of S-N curves in design requires consideration of their statistics.

4 Composite materials can be treated in the same way, but in marine applications the applied stresses are usually a very small proportion of the material strength and so fatigue damage is not a problem. Different fibres and matrix materials behave differently in fatigue, the properties of the matrix material being particularly important. In general, if fibres are aligned with the principal load then the S-N curves are very flat; stress concentrations can produce marked reductions in static strength but result in an even flatter S-N curve with fatigue life not being greatly affected. For further discussion of the advantages and disadvantages of marine structural materials see Chapter 5.

5 The S-N curve gives a design stress value (subject to scatter) for a particular number of stress cycles. Service conditions in ships involve a load spectrum of different stress ranges and these must be integrated to achieve a prediction of life. The conventional means of doing this follows Miner's (1945) hypothesis which enables groups of differing constant range stress cycles to be replaced by an equivalent number of cycles of a single magnitude. If a material is subjected to n_1 stress cycles at a stress amplitude σ_1 at which the expected fatigue endurance is N_1 cycles then it is assumed that the fraction n_1/N_1 of life has been used. If the stress is now changed to σ_2

for n_2 cycles then a further fraction n_2/N_2 of life is used. Hence for any number of stress levels failure will occur when

$$n_1/N_1 + n_2/N_2 + n_3/N_3 + \dots\dots\dots\dots = C \qquad (13.1)$$

where C ideally is equal to unity. However, as a safety factor, C is frequently made less than 1 to allow for scatter in the design curves, although in the method presented below for continuously varying stress cycles as are seen by a ship's hull, the curves themselves are modified to allow for the uncertainties in endurance limits (see paragraph 16).

6 The problem in the design of a ship's structure is therefore to ensure that the integrated number of stress cycles in all areas does not exceed the limit for the material during the ship's life. The greatest stress amplitudes will clearly occur at stress concentrations such as around holes and at the ends of superstructures, and it will be necessary to make as good an estimate as possible both of the stress under a given loading condition and the expected life in terms of number of wave encounters to assess whether the design is adequate.

Fatigue Design Practice

7 Structural efficiency and reliability should be critical factors in structural design and the cost and effort required to achieve a fatigue resistant structure should be balanced against the cost and loss of operational time involved in carrying out essential and unprogrammed structural repairs. There are two philosophies of fatigue design; the first, **fail-safe**, ensures that any local failure of a component or region of structure does not result in total structural failure, and that the loads are redistributed to other parts of the structure. The second, **safe-life**, means that local failure is not acceptable and the design must be such that no significant crack growth occurs in the design life of the structure. There is also a refinement of fail-safe design called **damage tolerant** design which uses fracture mechanics theory to check whether fatigue cracks will grow to be large enough to produce unacceptable failures before they can be detected during periodic inspections. This process is mainly applicable to structures such as pressure vessels and is unlikely to find applications in surface ships, because the complexity of the calculations required would not be justified.

8 Both these practices are relevant in different areas of ship structure, and in the main the primary structure should be designed for safe-life while other structure can be arranged to be fail-safe. Fail-safe structure in general will be lighter in weight but needs careful consideration of the intensity of stress fields and likely crack paths so that cracks do not progress from minor to major structure. On the other hand, even in primary structure it is possible to allow crack growth between adjacent holes provided that the remaining structure is sufficient *without* the material *between* the holes. However, the practical aspects should not be forgotten, that is that even if a crack is acceptable structurally, it may still allow ingress of water or other fluids to adjacent compartments and may also be a cause of accelerated corrosion.

9 Except where it is clearly uneconomic therefore, it is best to design ship structures using the safe-life philosophy, which should lead to a more robust structure with lower maintenance and repair costs. This means designing the primary structure with relatively low average field stresses as recommended in *Table 6.2* of Chapter 6, which then should allow for acceptable stress concentrations at the details discussed in the preceding chapters, without needing unreasonable effort on analysis to refine the design. Nevertheless, some local cracking must still be expected due to fabrication as well as design shortcomings. Thus it remains important, even with a safe-life philosophy, to use tough steels not susceptible to brittle failure under likely operational conditions.

Fatigue Design

10 From the foregoing discussion it will be clear that the process of fatigue design in surface ships involves first a suitable choice of average stress for the primary structure. The structure and its details are then designed in accordance with a set of rules which lead to the minimisation of stress concentrations, such as always providing adequate support to the ends of superstructures or always radiussing the corners of holes, and the stresses at those concentrations are calculated as discussed in the relevant chapters. It is then necessary to assess whether those stresses are acceptable taking account of the fatigue properties of the material and the expected load cycles in the ship's life.

11 A suitable procedure for this assessment is given by Clarke (1987) which is based on his proposed method of calculating the extreme loads in a ship's life presented in Clarke (1986) and in Chapter 3. The method is based on the assumption that at any point in the structure the stress range through hogging and sagging (or wave bending moment range) is approximately described by an exponential distribution, that is

(13.2) $$n(M) = n_L e^{-\alpha M}$$

where n (M) is the number of cycles exceeding a given bending moment M, n_L is the number of wave encounters for which the ship is to be designed and

(13.3) $$\alpha = -\{\ln (P_d/n_L)\}/M_d$$

with M_d being the design bending moment which has a probability P_d of being exceeded in n_L wave encounters. The justification for this assumption is given in the references.

12 For steel welded structures standard fatigue design curves are given in BS5400 Part 10 while, for aluminium, curves are given in BS8118 and for other materials in Boyer (1986). The steel curves, although intended for steel bridges, have been found in tests carried out by, or on behalf of, the Defence Research Agency, Dunfermline, to be representative of typical ship structures. Connection and detail geometries are grouped into eight classes which are illustrated in BS5400 Part 10 and with a description of structural details which are relevant to surface ships given in *Table 13.1* which assumes that all welds are proved sound and of good quality.

TABLE 13.1
Joint or Detail Fatigue Classes

Class		*Typical Examples*
B	a.	Plates free from welds with no flame cut edges or re-entrant corners.
	b.	Full penetration butt welds, parallel to direction of applied stress, with overfill dressed flush and finish machined.
C	a.	Unwelded plate flame cut by a controlled procedure to ensure cut surface is free from cracks.
	b.	Butt or fillet welds, parallel to direction of applied stress, made by an automatic submerged or open arc process with no undressed stop-start positions within the length.
	c.	Butt welds perpendicular to the direction of applied stress with overfill dressed flush.
D	a.	As C b. but containing stop-start positions.
	b.	Butt welds perpendicular to the direction of applied stress, made in the shop either manually or by an automatic process (except by submerged arc) providing all runs are in the flat (downhand) position.
E	a.	Butt welds perpendicular to the applied stress made on site either manually or by an automatic process.
	b.	Welded attachments to longitudinal stiffener webs.
F	a.	Full penetration butt joints, perpendicular to the direction of applied stress, made on a permanent backing strip or with differences in width or thickness of plate, machined to a smooth transition not steeper than 1:4.
	b.	Bevel-butt or fillet welded attachments to a stressed member; attachment length ⩽150 mm and distance from edge ⩾10 mm.
	c.	Load carrying fillet joint made with full penetration weld and with undercutting at corners dressed out by local grinding.
	d.	Weld attachments to longitudinal stiffener flanges, distance from flange edge ⩾10 mm.
F2	a.	Butt welded joints made between plates of unequal width but with weld ends ground to a radius not less than 1.25 times the plate thickness.
	b.	As F b. but with attachment length > 150 mm.
	c.	Parent metal adjacent to weld in a load carrying fillet joint made with partial penetration or fillet welds. Any undercutting at the corners of the member dressed out.
G	a.	Welded attachments to stressed members within 10 mm of edges or corners of stressed members.
	b.	Welded attachments to longitudinal stiffener flanges, distance from flange edge < 10 mm.
W		Weld metal in a load carrying joint made with fillet or partial penetration welds either transverse or parallel to the direction of applied stress (based on nominal shear stress on minimum weld throat area).

13 The fatigue endurance curves for the various classifications can all be described in the form

(13.4) $$N_f = A\,(\Delta\sigma)^{-b}$$

where N_f is the number of cycles of stress range $\Delta\sigma$ before failure and A and b are empirical constants. It is further assumed that welding residual stresses are such as to ensure that the local stress is always tensile so that the controlling parameter is $\Delta\sigma$ and that this is equal to M/Z for the normal assumption of simple beam theory for the hull.

14 From equation 13.2 it can be seen that the number of wave encounters N(M) giving a bending moment in the range M to M + δM is

(13.5) $$N(M) = \alpha\, n_L e^{-\alpha M}\,\delta M$$

and if equations 13.4 and 13.5 are substituted in equation 13.1 Miner's rule can be re-expressed as

$$\int_0^\infty \frac{\alpha n_L e^{-\alpha M}}{A(M/Z)^{-b}}\, dM = \frac{n_L \Gamma(b+1)}{A(\alpha Z)^b} = 1.0 \qquad (13.6)$$

and noting that if the permissible design stress range is σ_d, $Z = M_d/\sigma_d$, and so

$$\sigma_d = \{\ln(P_d/n_L)\}\ \{A/[n_L\Gamma(b+1)]\}^{1/b} \qquad (13.7)$$

Hence for a given life and probability of exceedance, and for a given class of detail, the allowable stress range in steel or other metals can easily be calculated.

15 If the recommended figures from Chapter 3 of $P_d = 0.01$ and $n_L = 3 \times 10^7$ are inserted into equation 13.7 then the value of σ_d depends on material and class of detail alone. However, the exponential distribution for the probability of a stress range does not include any allowance for slam induced whipping, and Clarke (1987) deduces from records of strains taken during slamming trials that the allowable stress should be reduced by 5% for frigate and destroyer type ships, although this could be as much as 80% for flat bottomed vessels such as landing craft which slam on almost every wave. Taking account of these factors therefore (including a 5% reduction), the values of A and b, and the allowable stresses for different joint classifications in frigates and destroyers, are given in *Table 13.2* for steel and aluminium.

TABLE 13.2
Fatigue *Design* Criteria

Design Detail	*Fatigue Curve Constants (Steel)*		*Permissible Stress Range (MPa)*	
Class	A	b	*Steel*	*Aluminium*
B	1.01×10^{15}	4.0	716	239
C	4.23×10^{13}	3.5	589	196
D	1.52×10^{12}	3.0	424	141
E	1.04×10^{12}	3.0	373	124
F	0.63×10^{12}	3.0	316	105
F2	0.43×10^{12}	3.0	278	93
G	0.25×10^{12}	3.0	232	77
W	0.16×10^{12}	3.0	200	67
Parent Plate				
Steel BS4360				
grade 43A/43D	1.88×10^{30}	10	874	
grade 50EE	3.10×10^{23}	7	1193	
Aluminium Alloy				
5083/O	3.80×10^{62}	26		259

16 The constants in *Table 13.2* assume the Miner's summation adding to unity, but the fatigue endurance curves being adjusted *downwards* to the mean less two standard deviations to cover the scatter in results. The results of this process have been validated for the uptake and downtake openings in the upper deck of WHITBY and ROTHESAY Class frigates where a class problem became apparent; all ships of the classes exhibited cracks running from one or both of the holes at around 20 years life. The reference analyses the structure using finite elements to establish the stress pattern in the area and then checks the result against the allowable stresses. In this case, to achieve a proper comparison, the mean values of the constants A and b have been used (given for information in *Table 13.3*), and within the broad spectrum that would be expected, it was shown that had the method been available when the ships were designed, fatigue cracking at about 15 to 20 years would have been predicted.

Table 13.3
Mean Fatigue Curve Constants

Design Detail	*Fatigue Curve Constants (Steel)*	
Class	A	b
B	2.34×10^{15}	4.0
C	1.08×10^{14}	3.5
D	3.99×10^{12}	3.0
E	3.29×10^{12}	3.0
F	1.73×10^{12}	3.0
F2	1.23×10^{12}	3.0
G	0.57×10^{12}	3.0
W	0.37×10^{12}	3.0
Parent Plate		
Steel BS4360		
grades 43A/43D	4.91×10^{30}	10
grade 50EE	8.90×10^{23}	7
Aluminium Alloy		
5083/O	9.92×10^{62}	26

17 The figures for the constants for the classes of detail in *Tables 13.2* and *13.3* assume a stress ratio R = 0, defined as the ratio of the minimum to maximum stress in the cyclic range. That is, as mentioned in paragraph 14, the stress range is assumed entirely tensile and the minimum stress is zero. For the parent plate, however, a symmetrical tensile to compressive range is more appropriate as residual stresses are small, and the constants in the tables are for R = -1.

18 For the procedure described above to be valid it is essential that the extent of plasticity in the region surrounding the stress concentration is limited, and to ensure that this is so the average stress in the structure should not exceed 75% of the yield stress in either hogging or sagging. In practice the average stress should be much less than this value if the stress ranges in *Table 13.2* have any chance of being met, and a maximum stress *range*, under the extreme loads defined in Chapter 3, of about 300 MPa in steel is probably a good aim during design.

19 It should be noted here also that the stresses which must be derived in any analysis are those at the stress concentration, but *not* the stress due to a weld profile or at a crack tip. The validation above is sufficient for holes in a structure but the method is not yet fully validated for stress concentrations in other areas such as around the ends of superstructures. In this case the stress required is that near to the toe of the weld at the junction between the deck and superstructure end, which is very difficult to find even using finite element analysis; in fact in an idealised structure the stress in infinite, and so refining the finite element mesh gives progressively higher stresses which are meaningless. There are two approximations which can be used in this situation. The first is based on the fact that the design fatigue curves which are appropriate to a superstructure end (Class F) were derived from tests on specimens with stress concentrations as illustrated in *Figure 13.2*, and the reference stress used was not the stress local to the stress concentration but the average stress applied to the specimen. This corresponds to the stress about 10 plate thicknesses away from the stress concentration. It is therefore appropriate to enter the Class F curve with the stress at about this distance from a superstructure corner. An alternative approach is to use the so-called 'hot spot' stress method as recommended in design rules for node joints in offshore structures. This relates the stress at the likely point of failure, that is the toe of the weld, to a 'hot spot' stress fatigue design curve which is close to that for Class D. In calculating the 'hot spot' stress no account is taken of the weld geometry, and in an idealised finite element representation (ignoring the weld) the stress is approximately equal to the value at about one plate thickness from the corner. Current advice from Hugill and Sumpter (1989(a)) is that the 'hot spot' stress for entry to the Class D curve should be taken at two thirds of the plate thickness from the corner. In any case, if there is any doubt as to the acceptability of such a joint under fatigue loading then specialists should be consulted, but if the rules for structural continuity given in Chapter 8 are followed, then the problem should at least be contained.

Design Example

20 A brief example illustrating the use of the joint classes and allowable stresses may be useful. Taking the upper deck structure from the example in Chapter 7, the design sagging stress is 149.1 MPa (from *Table 7.4*); using the design bending moments from paragraph 3 of Chapter 7 the design hogging stress is therefore 149.1 × 320/400 = 119.3 MPa and so the stress range is 268.4 MPa.

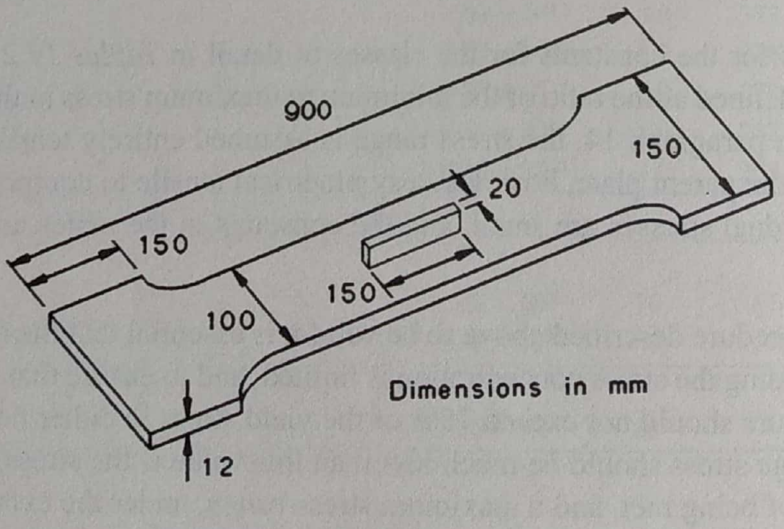

Figure 13.2 Class F Fatigue Specimen

21 If a flame cut hole with no welds in its vicinity exists in this deck, it is classified C in *Table 13.1* and from *Table 13.2* the permissible stress range is 589 MPa. Reinforcement to the hole must therefore be designed such that the stress concentration factor is restricted to 589/258.4 = 2.2.

22 Any butt welds across the deck will come under Classes D, E, F or F2 depending on how the weld is finished and whether there are steps in thickness or width. With the stress range of 268.4 MPa, inspection of *Table 13.2* shows that welds to Classes D, E and F are acceptable but that Class F2 welds in the deck would not be.

Reliability Considerations

23 Because of the statistical nature of the results achieved from any fatigue tests it is natural that the use of probability theory should have been considered to assist the designer, one of the earliest proposals being due to Orbeck (1969) who related the probability of failure to the number of overstressed crystals in the material. In subsequent years much research work is presented in the reports of the International Ship and Offshore Structures Congress, see, for example, the report of Committee III.2 (1988) which includes a large number of references and in particular draws attention to Wirsching and Wu (1985) for a summary of progress and later information is presented by Wirsching and Chen (1988).

24 For convenience in analysis it is usually assumed that the value of b in equation 13.4 is a constant for a particular material and joint configuration, but that the value of A is a random variable accounting for all the uncertainty in the process, and generally taken to be log-normal. Consequently the shift in the fatigue endurance curve from the mean to a design level (discussed in paragraph 17 and reflected in *Table 13.2*) can be defined in terms of b and of the mean and standard deviation of $\log_{10}A$, and the design level, as described above, is commonly taken as two standard deviations below the mean. This implies that about 2.5% of the data will fall below the design curve.

25 There are, however, other uncertainties beyond the material test data, the first being lack of information at the low stress, high cycle end of the curve, which is ill-defined in most cases. Additionally, there are uncertainties in service life and the environment met over that time, on the estimation of stress concentrations, and not least the use of Miner's rule. Quantification of these uncertainties has involved much effort in recent years but still remains mostly in the research arena; the best information available for the use of designers may be found in White and Ayyub (1987), also in Madsen (1986) who discusses the use of the results of periodic inspections to update fatigue life predictions and so circumvent the uncertainties.

26 It should be noted also that the value of α in equation 13.3 contains three parameters, P_d, n_L and M_d, all of which involve considerable uncertainties, and which are therefore carried over into the acceptable stresses given in *Table 13.2*. In any particular design case, if a more reliable value of α can be substantiated, then the acceptable stresses should be modified accordingly.

Fatigue Crack Growth

27 If, in spite of all precautions, fatigue cracks do develop, their likely rate of extension can be predicted by fracture mechanics. The basic tool for this analysis is the linear elastic stress intensity factor, K. Exact values of K for any given geometry can be calculated by finite element analysis, but as a quick estimate of crack growth rate a simple approximation to K may suffice. For instance, for a crack of length 2a in a structure subjected to a uniform tensile stress σ, the stress intensity factor is given by

$$K = \sigma\sqrt{(\pi a)} \qquad (13.8)$$

The rate of fatigue crack extension Δa (from each crack tip) per cycle of wave loading may then be estimated from

$$\Delta a = 1.47 \times 10^{-7} \, (\Delta K)^{2.7} \qquad (13.9)$$

where

$$\Delta K = \Delta\sigma\sqrt{(\pi a)} \qquad (13.9a)$$

and Δσ is the stress range normal to the crack in MPa, a is the half crack length in metres and Δa is given in millimetres. The constant is taken from Table 3 of Booth and Dobbs (1986) for a seawater environment upper 95% confidence limit.

28 Δσ is the stress range at the location of the crack in the crack free structure. If Δσ varies significantly over the length of the crack, for example a crack propagating out from a corner of a hole, equation 13.9a can be amended to be

$$\Delta K = [(3\Delta\sigma_m + \Delta\sigma_t)/4] \, \sqrt{(\pi a)} \qquad (13.9b)$$

where $\Delta E\sigma_m$ is the average stress range over the length of the crack and $\Delta E\sigma_t$ is the stress range at the location of the crack tip (in the absence of the crack).

29 *Table 13.4* provides the expected spectrum of wave loadings in a life of 3×10^7 wave encounters derived from measurements at sea as discussed in Chapter 3, and presented here by stress level and number of cycles. To estimate crack growth rate calculate Δa for each of these stress levels and multiply the result by the corresponding number of cycles. The sum of the individual products of Δa and n cycles is the expected total crack growth in the ship's life. The expected crack growth in a shorter period can be obtained as a direct proportion of this value, but of course the calculation will be non-conservative if the ship encounters disproportionately heavy weather in the period under consideration. There is also some non-conservatism because this simple calculation does not update the crack geometry. If more precise estimates are needed of crack growth rate then specialists should be consulted.

TABLE 13.4
Spectrum of Stress Range Cycles

$\Delta\sigma/\Delta\sigma_d$	Number of cycles
0.025	19.04×10^6
0.075	7.41×10^6
0.125	2.35×10^6
0.175	8.02×10^5
0.225	2.57×10^5
0.275	9.60×10^4
0.325	2.96×10^4
0.375	1.01×10^4
0.425	3.39×10^3
0.475	1.07×10^3
0.525	359
0.575	131
0.625	40
0.675	14
0.725	5
0.775	2
0.825	1

($\Delta\sigma_d$ is the stress range due to the design bending moment range, hog to sag)

Fracture

30 As noted in paragraph 2, brittle fracture is unlikely under normal wave loading, but in view of the serious consequences of extensive unstable crack propagation, every precaution must be taken to avoid its possible occurrence. The most effective defence is to use steels and weld procedures with certified toughness at all locations where brittle fracture would be dangerous. This essentially means specifying defect tolerant materials for the whole outer hull envelope of the ship. On present evidence the use of steel to BS4360 grade 43D or better will avoid significant risk of brittle fracture in plate less than 25 mm thick, but care should be taken to ensure that all weld procedures at least match the parent plate in toughness. At plate yield stress levels of not more than 450 MPa this should not be a problem, but particular attention should be paid to any new proposed procedures involving a high deposition rate of weld metal. Such innovations should always be subjected to weld procedure testing reproducing exactly the welding parameters to be used in the ship. Charpy crystallinity at 0°C should be less than 50%. In important cases Charpy testing should be supplemented by tests to measure the critical toughness factor K_{JC} at the appropriate strain rate and temperature.

Note that K_{JC} should not be confused with K_{IC}, the critical plane strain fracture toughness. K_{JC} is the elasto-plastic toughness relating to a specific set of structural conditions.

31 The following parameters are recommended for determination of K_{JC} using procedures described more fully by Sumpter et al (1989). The minimum required toughness under these conditions is such that $K_{JC} \geqslant 150$ MPa $\sqrt{m}$.

Test temperature	0°C
Rate of increase of K	3×10^4 MPa $\sqrt{m}/s$
Thickness of specimen	As structural thickness
Geometry	Three point bend test
Dimensions of specimen	Width = 50 mm Span = 200 mm Crack depth = 25 mm.

32 To determine whether there is any risk of brittle fracture from a crack found in service it is first necessary to ascertain the type of material adjacent to the crack tip. If the plate material is to BS4360 grade 43D or equivalent or better (see Chapter 5) and is less than 25 mm thick there is no significant risk. If the crack is close to a weld but is in plate material, including the heat affected zone, there is still little risk. If the crack appears to be in the weld itself then safety is less assured because of sparsity of data on weld properties. In this case, or if the plate material is to BS4360 grade 43A or equivalent with no certified toughness, then the following procedure should be used.

33 Calculate the elastic stress intensity factor, either exactly using finite element analysis, or approximately from equation 13.8 using an applied stress of 100 MPa if the crack is remote from a weld, or 400 MPa if the crack is adjacent to a weld (to allow for residual stress). If the value of K so derived exceeds 150 MPa $\sqrt{m}$ there will be some risk of brittle fracture under adverse conditions, for example storm conditions combined with low temperatures. A temporary repair of the crack should be made by drilling a hole at its tip as described in paragraph 36.

Survey and Repair

34 Survey and repair of structure is discussed in more detail in Chapter 16 and in NES155 Part 2 for RN ships, but a few comments are appropriate in the context of fatigue cracks. In surveying a structure, the surveyor should always have in mind the possibility of fatigue cracking at any structural discontinuity, such as around the ends of superstructures and especially the bridge front, and at the corners of any holes in main structure. Cracks are also likely in minor structure where this is stressed locally by main structure, for example in minor bulkheads below the bridge front.

35 Cracks discovered in primary structure should always be repaired, and if the ship has a significant period of life remaining then the repair should be undertaken by cutting out the complete fatigue damaged area and replacing it with new plate. The extent of replacement will depend on circumstances, but should include plate around all significant stress concentrations in the vicinity even if cracking has not yet commenced at some of them. Repair by fitting doubler plates is never satisfactory, as it creates a joint with a much worse fatigue classification and only serves to cover up the damage, which will continue to grow; such a repair is only suitable for providing watertightness in a ship with little remaining life. The only exception to this rule is where the doubler is large and significantly stiffer than the parent plate, for example when a steel or carbon fibre patch is fitted on aluminium. The effect then is to take the load out of the fatigue damaged area, but the fixity of the doubler is of especial importance with the loads in the connections to the parent plate being very large. Suitable processes are described by Allen et al (1988).

36 If an immediate repair is not feasible then the best approach is to locate the crack tip (by magnetic particle detection or some other reasonably reliable method, as the crack tip is rarely obvious to the naked eye), and to drill a hole through to eliminate the crack tip. The hole should have a diameter the same or a little larger than the plate thickness. The performance will then be improved if a steel bolt is fitted into the hole and torqued up so that it is almost yielded; the effect is to put the plate locally into high compression through the thickness so that any applied tensile load does not cause large effective tensile stresses in the region of the defect. Using this procedure, the fatigue life locally can be increased by up to a factor of four as described by Hugill and Sumpter (1989(b)), as well as having the advantage of restoring watertightness.